

Performance optimization of a solar driven heat engine with finite-rate heat transfer

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Abstract

An optimal performance analysis for an equivalent Carnot-like cycle heat engine of a parabolic-trough direct-steam-generation solar driven Rankine cycle power plant at maximum power and maximum power density conditions is performed. Simultaneous radiation-convection and only radiation heat transfer mechanisms from solar concentrating collector, which is the high temperature thermal reservoir, are considered separately. Heat rejection to the low temperature thermal reservoir is assumed to be convection dominated. Irreversibilities are taken into account through the finite-rate heat transfer between the fixed temperature thermal reservoirs and the internally reversible heat engine. Comparisons proved that the performance of a solar driven Carnot-like heat engine at maximum power density conditions, which receives thermal energy by either radiation-convection or only radiation heat transfer mechanism and rejects its unavailable portion to surroundings by convective heat transfer through heat exchangers, has the characteristics of (1) a solar driven Carnot heat engine at maximum power conditions, having radiation heat transfer at high and convective heat transfer at low temperature heat exchangers respectively, as the allocation parameter takes small values, and of (2) a Carnot heat engine at maximum power density conditions, having convective heat transfer at both heat exchangers, as the allocation parameter takes large values. Comprehensive discussions on the effect of heat transfer mechanisms are provided.

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Nomenclature

$\dot{Q}$	heat transfer rate (W)
$\bar{Q}$	non-dimensional heat transfer rate
P	pressure (Pa)
R	ideal gas constant (kJ/kg K)
s	entropy (J/kg K)
T	temperature (K)
V	volume (m ³)
$\dot{W}$	power (W)
$\bar{W}$	non-dimensional power

Greek

α	heat transfer coefficient
β	allocation parameter
η	thermal efficiency (%)
τ	T_L/T_H
χ	T_W/T_H

Subscripts

C	Carnot; cold
c	convective
CNCA	Chambadal-Novikov-Curzon-Ahlborn
d	density in power density concept
H	related to high temperature thermal reservoir
L	related to low temperature thermal reservoir
max	maximum
min	minimum
mp	at maximum power conditions
mpd	at maximum power density conditions
opt	optimum
r	radiative
c–c	convective heat transfer at both thermal reservoirs
r–c	radiative heat transfer at hot side and convective heat transfer at cold side reservoirs
r–r	radiative heat transfer at both thermal reservoirs
W	warm
4	state point at the end of the isentropic expansion process shown in Fig. 3

1. Introduction

Electricity generation by solar power plants has gained importance in recent years. Currently, three concepts are well known and established: Parabolic trough power plants, solar tower power plants and Dish/Stirling systems. Parabolic trough power plants which

are also referred to as solar farm power plants use parabolic reflectors in a trough configuration and are the most mature solar thermal technology. Troughs concentrate the solar heat flux up to 100 times on a fluid-filled receiver tube positioned along the line of focus in the trough. A synthetic heat transfer fluid (HTF) heated up to 400 °C in the collector field transfers heat to the working fluid in a heat exchanger then returns to the collector field around 300 °C. Therefore, a parabolic trough solar power plant with HTF employs two circuits.

Recently, the subject of major research has been focused on direct steam generation (DSG) in the absorber pipe as an ideal solution in order to make parabolic trough power plants more competitive, more efficient and at less cost since this system does not need a number of plant components, such as heat exchangers and oil pumps which consume high power. Basic concepts of DSG are explained in detail in [1–4]. A simplified schematic diagram of a parabolic trough solar driven Rankine cycle power plant with DSG is illustrated in Fig. 1.

To carry out an optimal performance analysis for such a system, its ideal Rankine cycle, which is shown in Fig. 2, is modified to an equivalent Carnot-like cycle with finite-rate heat transfer employing a similar approach as in [5,6]. An entropy averaged temperature, i.e. the amount of heat added to the Rankine cycle divided by the entropy change experienced during the heat addition process, is defined as $T_w = \Delta Q / \Delta s$ for the hot temperature of the working fluid in the corresponding Carnot-like cycle.

Although, the thermal efficiency of a Carnot cycle gives an upper bound for thermal efficiency, it does not have great significance and is a poor guide to the performance of real heat engines [7]. In order to analyze of a certain amount of power production with finite size devices, Chambadal [8], Novikov [9] and Curzon-Ahlborn [10] extended the ideal Carnot cycle heat engine model, in which any temperature difference between working fluid and thermal reservoirs is not considered, to their own Carnot cycle heat engine models by taking finite temperature difference between working fluid and thermal reservoirs into account and investigated the performances of their heat engine models at

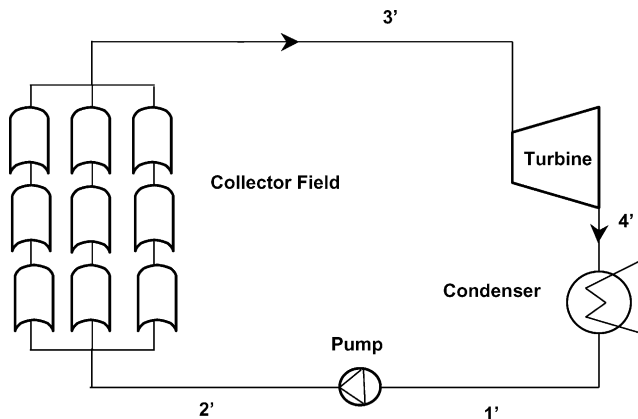


Fig. 1. Simplified schematic diagram of a parabolic trough solar power plant with DSG.

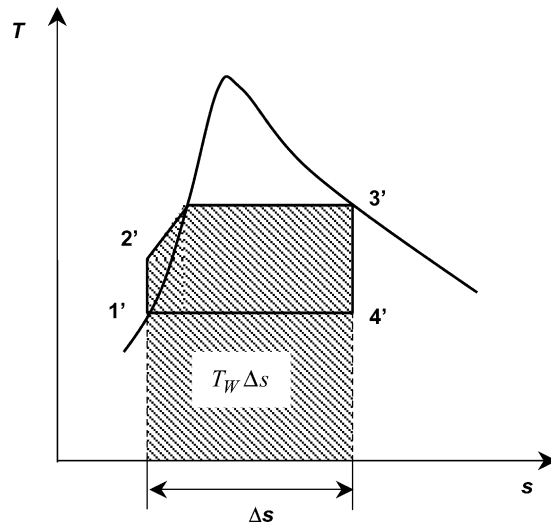


Fig. 2. T-s diagram of a simple Rankine heat engine cycle.

maximum power (mp) conditions. They derived independently that either the hot end temperature T_W or the difference between hot and cold end temperatures $T_W - T_C$ of a power plant can be optimized to maximize the power output. These pioneering researchers obtained the efficiency of their Carnot heat engine at mp conditions as

$$\eta_{mp} = 1 - \sqrt{\frac{T_L}{T_H}} \quad (1)$$

taking only convective heat transfer at both heat exchangers into account together with the second law of thermodynamics as a constraint. This efficiency was referred to the Chambadal-Novikov-Curzon-Ahlborn efficiency or briefly the CNCA efficiency (η_{CNCA}) by Bejan [11].

Research studies have been carried out for solar driven heat engines and solar collectors to present the principle of operation of a solar thermal power plant considering a heat engine having finite temperature difference [12,13]. The CNCA efficiency depends only on the temperatures of the hot and cold thermal reservoirs between which the heat engine operates. It is reported that for radiative heat transfer at both hot and cold thermal reservoir sides, the thermal efficiency depends on the radiation heat transfer coefficients at both thermal reservoirs [14]. To include the effect of engine size on the performance analysis, a performance analysis criterion called maximum power density (mpd) was defined as the ratio of power to the maximum specific volume in the cycle by Sahin et al. [15]. An analysis using mpd criterion for a Carnot heat engine having finite temperature difference has also been performed to conclude that the thermal efficiency at mpd conditions is always greater than the thermal efficiency at mp conditions [16]. An analysis for a solar driven Carnot heat engine having finite temperature difference working at mp conditions to investigate the collective role of the radiation heat transfer from the high temperature reservoir and convection heat transfer to the low temperature reservoir was performed in [17].

The objective of the present study is to investigate the optimal performance of a simplified parabolic trough solar power plant with DSG considering an equivalent solar driven Carnot-like heat engine having finite temperature difference between working fluid and thermal reservoirs. The effects of radiation only and simultaneous radiation-convection heat transfer from the high temperature thermal reservoir to the working fluid together with convection heat transfer from the working fluid to the low temperature thermal reservoir on the performance and design parameters of this heat engine working at mp and mpd conditions are particularly studied in detail.

2. The theoretical model

The model of the solar driven Carnot-like heat engine working at mpd conditions having finite temperature difference considered in this study is shown in Fig. 3. The hot and cold end working fluid temperatures are illustrated as T_W and T_C in Fig. 3.

In this model, heat transfer from the working fluid to the cold thermal reservoir is assumed mainly by convection whereas heat transfer from the hot thermal reservoir to the working fluid is simultaneously by radiation and convection which can be expressed as

$$\dot{Q}_L = \alpha_{L,c}(T_C - T_L) \quad (2)$$

$$\dot{Q}_H = \alpha_{H,r}(T_H^4 - T_W^4) + \alpha_{H,c}(T_H - T_W) \quad (3)$$

respectively. According to the first law of thermodynamics, the power output for this heat engine is

$$\dot{W} = \dot{Q}_H - \dot{Q}_L = \eta \dot{Q}_H. \quad (4)$$

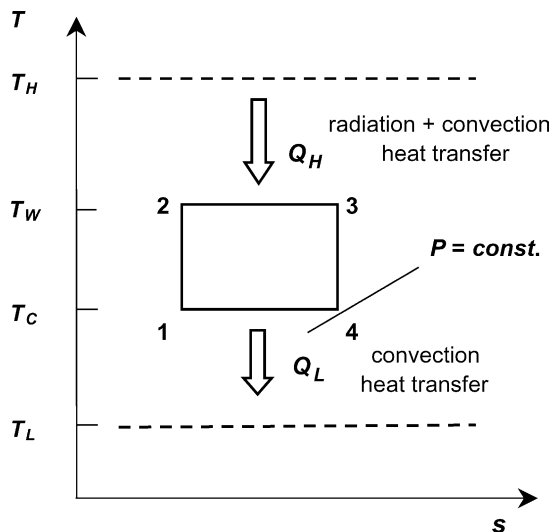


Fig. 3. Simplified solar-driven Carnot heat-engine model.

The second law of thermodynamics for this Carnot heat engine having finite temperature difference is also written as

$$\frac{\dot{Q}_H}{T_W} = \frac{\dot{Q}_L}{T_C}. \quad (5)$$

Substitution of Eqs. (2) and (3) into Eq. (5) yields a relationship between T_W and T_C as

$$T_C = \frac{T_L}{1 - \frac{\alpha_{H,r}}{\alpha_{L,c}} \left(\frac{T_H^4 - T_W^4}{T_W} \right) - \frac{\alpha_{H,c}}{\alpha_{L,c}} \left(\frac{T_H - T_W}{T_W} \right)}. \quad (6)$$

Then, the power output from Eq. (4) becomes

$$\dot{W} = \left(1 - \frac{T_C}{T_W} \right) [\alpha_{H,r}(T_H^4 - T_W^4) + \alpha_{H,c}(T_H - T_W)]. \quad (7)$$

In the present study, the power density is also used as an objective function which is defined as the power output divided by the maximum volume in the cycle

$$\dot{W}_d = \frac{\dot{W}}{V_{\max}} \quad (8)$$

where $V_{\max} = mRT_C/P_{\min}$, which is shown as $V_{\max} = V_4$ in Fig. 3, assuming that the working fluid is an ideal gas. The minimum pressure in the cycle, which is also shown as $P_{\min} = P_4$ in Fig. 3, is taken to be constant.

By defining the non-dimensional allocation parameters of $\beta_{r-c} = \alpha_{H,r}T_H^3/\alpha_{L,c}$, $\beta_{c-c} = \alpha_{H,c}/\alpha_{L,c}$ regarding to the radiation-convection and convection-convection heat transfer mechanisms at hot-cold side heat exchangers, respectively and the temperature ratios of $\tau = T_L/T_H$ and $\chi = T_W/T_H$, the thermal efficiency ($\eta = 1 - (T_C/T_W)$), the non-dimensional power output ($\bar{W} = \dot{W}/\alpha_{L,c}T_H$), the non-dimensional power density ($\bar{W}_d = \dot{W}_d/(\alpha_{L,c}P_{\min}/mR)$) and the non-dimensional heat transfer rate from the hot thermal reservoir to the working fluid ($\bar{Q}_H = \dot{Q}_H/\alpha_{L,c}T_H$) for the solar driven Carnot heat engine having finite temperature difference can be expressed, respectively as follows

$$\eta = 1 - \frac{\tau}{\chi - \beta_{r-c}(1 - \chi^4) - \beta_{c-c}(1 - \chi)}, \quad (9)$$

$$\bar{W} = \bar{Q}_H\eta = [\beta_{r-c}(1 - \chi^4) + \beta_{c-c}(1 - \chi)]\eta, \quad (10)$$

$$\bar{W}_d = \frac{[\beta_{r-c}(1 - \chi^4) + \beta_{c-c}(1 - \chi)][\chi - \beta_{r-c}(1 - \chi^4) - \beta_{c-c}(1 - \chi) - \tau]}{\chi\tau} \quad (11)$$

To find the non-dimensional maximum power and power density for this solar driven heat engine, we differentiated Eqs. (10) and (11) with respect to χ , set the resultant

derivative equal to zero (i.e. $\partial \bar{W}/\partial \chi = 0$ and $\partial \bar{W}_d/\partial \chi = 0$, respectively) and obtained

$$\begin{aligned} & 4\beta_{r-c}^3\chi^{11} + \beta_{r-c}^2(9\beta_{c-c} + 8)\chi^8 - 8\beta_{r-c}^2(\beta_{c-c} + \beta_{r-c})\chi^7 \\ & + 2\beta_{r-c}(3\beta_{c-c}^2 + 5\beta_{c-c} + 2)\chi^5 - \beta_{r-c}[2(\beta_{c-c} + \beta_{r-c})(5\beta_{c-c} + 4) + 3\tau]\chi^4 \\ & + 4\beta_{r-c}(\beta_{c-c} + \beta_{r-c})^2\chi^3 + \beta_{c-c}(\beta_{c-c} + 1)^2\chi^2 - 2\beta_{c-c}[(\beta_{c-c} + \beta_{r-c})(\beta_{c-c} + 1)]\chi \\ & + (\beta_{c-c} + \beta_{r-c})[\beta_{c-c}(\beta_{c-c} + \beta_{r-c}) - \tau] = 0 \end{aligned} \quad (12)$$

and

$$\begin{aligned} & 7\beta_{r-c}^2\chi^8 + \beta_{r-c}(8\beta_{c-c} + 4)\chi^5 - 3\beta_{r-c}[2(\beta_{c-c} + \beta_{r-c}) + \tau]\chi^4 \\ & + \beta_{c-c}(\beta_{c-c} + 1)\chi^2 - (\beta_{c-c} + \beta_{r-c})(\beta_{c-c} + \beta_{r-c} + \tau) = 0. \end{aligned} \quad (13)$$

We solved Eqs. (12) and (13) numerically and obtained the non-dimensional maximum power output $\bar{W}_{\max}$, the non-dimensional maximum power density $\bar{W}_{d,\max}$ and the corresponding efficiencies of η_{mp} and η_{mpd} using the physically meaningful roots of χ which take place between 0 and 1.

The significance of Eqs. (12) and (13) is that their real roots provide the optimum design parameters of the solar driven Carnot heat engine working at mp and mpd conditions for any given τ , β_{r-c} and β_{c-c} values.

Eqs. (9)–(11) define the generalized performance functions of the solar driven Carnot heat engine and Eqs. (12), (13) indicate the optimum conditions where the power output and power density are maximized, respectively. We now consider and examine two special cases in detail to demonstrate that Eqs. (12) and (13) have generalized characteristics.

Case 1. When $\beta_{r-c} = 0$, the heat engine is a Carnot heat engine having finite-rate convective heat transfer with both of the thermal reservoirs then Eq. (12) becomes

$$(\beta_{c-c} + 1)^2\chi^2 - 2\beta_{c-c}(\beta_{c-c} + 1)\chi + (\beta_{c-c}^2 - \tau) = 0 \quad (14)$$

and the optimal value of χ can be obtained as

$$\chi_{\text{opt}} = \frac{\beta_{c-c} + \sqrt{\tau}}{1 + \beta_{c-c}}. \quad (15)$$

Hence, the non-dimensional power output and the corresponding efficiency can be found as

$$\bar{W}_{\max} = \frac{\beta_{c-c}(\sqrt{\tau} - 1)^2}{1 + \beta_{c-c}} \quad (16)$$

and

$$\eta_{\text{mp}} = 1 - \sqrt{\tau}. \quad (17)$$

Eqs. (16) and (17) are the well known results for the Carnot heat engine working at mp conditions having finite-rate heat transfer with the thermal reservoirs.

For the case considered, Eq. (13) becomes

$$\beta_{c-c}(\beta_{c-c} + 1)\chi^2 - \beta_{c-c}(\beta_{c-c} + \tau) = 0 \quad (18)$$

and the optimal value of χ is obtained

$$\chi_{\text{opt}} = \frac{\sqrt{\beta_{c-c} + \tau}}{\sqrt{1 + \beta_{c-c}}}. \quad (19)$$

Then, the non-dimensional power density and the corresponding efficiency are found as

$$\bar{W}_{d,\text{max}} = \frac{\beta_{c-c}}{\tau} \left(1 + 2\beta_{c-c} + \tau - 2\sqrt{1 + \beta_{c-c}}\sqrt{\beta_{c-c} + \tau} \right) \quad (20)$$

and

$$\eta_{\text{mpd}} = 1 - \frac{\tau}{\sqrt{(1 + \beta_{c-c})(\beta_{c-c} + \tau)} - \beta_{c-c}}. \quad (21)$$

It may easily be proven that Eqs. (20) and (21) are equivalent to those reported by Kodál et al. [18].

Case 2. When $\beta_{c-c} = 0$, the heat engine is a solar driven Carnot heat engine with finite-rate heat transfer by radiation only at high temperature thermal reservoir and by convection at low temperature thermal reservoir. Eqs. (12) and (13) then become.

$$4\beta_{r-c}^2\chi^{11} + 8\beta_{r-c}\chi^8 - 8\beta_{r-c}^2\chi^7 + 4\chi^5 - (8\beta_{r-c} + 3\tau)\chi^4 + 4\beta_{r-c}^2\chi^3 - \tau = 0 \quad (22)$$

and

$$7\beta_{r-c}\chi^8 + 4\chi^5 - (6\beta_{r-c} + 3\tau)\chi^4 - (\beta_{r-c} + \tau) = 0. \quad (23)$$

Eqs. (22) and (23), which can only be solved numerically, were also derived by Sahin [17,19] and Koyun [20]. Therefore our generalized functions of Eqs. (12) and (13) cover both these special cases.

3. Results and discussion

A numerical study is carried out to determine the effects of allocation parameters β_{r-c} , β_{c-c} and the reservoir temperature ratio τ on the design parameter χ and on the thermal efficiency at mpd conditions η_{mpd} . Comparisons for the optimal design parameters of the solar driven Carnot heat engines considered in this study working at mp conditions and at mpd conditions are provided. Special case 2 mentioned in Section 2 is also studied with comparisons in detail. In this parametric study, the temperature of the low temperature reservoir, T_L is assumed to be fixed at 300 K.

The variation of χ with respect to the radiation-convection allocation parameter β_{r-c} for three cases of temperature ratio $\tau = T_L/T_H$ when the convection-convection allocation parameter $\beta_{c-c} = 0.1$ is shown in Fig. 4a. It is observed from Fig. 4a that the optimal temperature ratio $\chi = T_W/T_H$ increases and approaches to 1 as β_{r-c} increases for both mp and mpd conditions, since β_{r-c} is proportional to the radiative heat transfer coefficient at hot side to the convective heat transfer coefficient at cold side and its increase causes T_W approach to T_H . For higher values of τ , i.e. as T_H approaches to T_L , χ always takes greater values which means that T_W approaches to T_H again however now it is valid for each value of β_{r-c} considered.

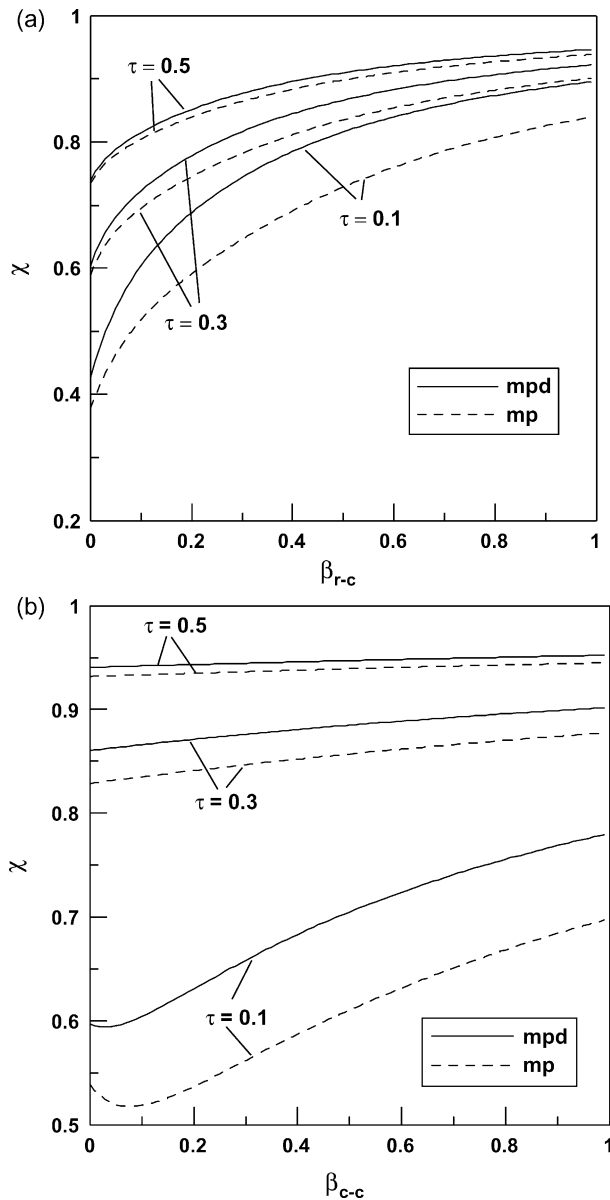


Fig. 4. Effect of the allocation parameters (a) β_{r-c} and (b) β_{c-c} on the optimal temperature ratio $\chi = T_W/T_H$ for three cases of τ when $\beta_{c-c}=0.1$ and $\beta_{r-c}=0.1$, respectively.

Fig. 4b shows the effect of the allocation parameter β_{c-c} on the optimal temperature ratio χ for three cases of τ and a fixed value of $\beta_{r-c}=0.1$. For small values of τ , i.e. $T_H \gg T_L$, χ shows a weak minimum for small values of β_{c-c} and increases for large values of β_{c-c} . As τ takes higher values, i.e. $T_H \rightarrow T_L$, χ also takes high values for each β_{c-c} . As τ

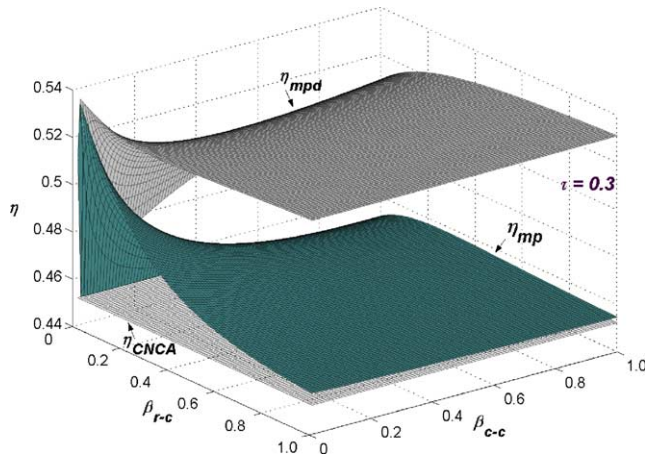


Fig. 5. Variation of efficiency η for the solar driven Carnot heat engine working at mp and mpd conditions with respect to the allocation parameters of β_{r-c} and β_{c-c} for $\tau=0.3$.

increases, the rate of increase in χ with respect to β_{c-c} gets smaller and the effect of the allocation parameter β_{c-c} is insignificant in this case since the difference between T_H and T_W also gets smaller.

It is also observed from Fig. 4a and b that χ always takes higher values for the solar driven Carnot heat engine working at mpd conditions than that of at mp conditions for each value of β_{r-c} and β_{c-c} considered, however the difference gets smaller as τ increases.

The variations of efficiencies η for the solar driven Carnot heat engine working at mp and mpd conditions with respect to the allocation parameters of β_{r-c} and β_{c-c} for $\tau=0.3$ are illustrated in Fig. 5. It is shown in Fig. 5 that thermal efficiency at mpd conditions is always higher than that of at mp conditions for all realistic range of allocation parameters of β_{r-c} and β_{c-c} . As β_{r-c} and β_{c-c} have high values, the change in both efficiencies becomes smaller.

To understand better the effect of β_{r-c} and β_{c-c} separately, variations of efficiency with respect to β_{r-c} and β_{c-c} are analyzed while holding one of them fixed at three different values for mp and mpd conditions. Variations of efficiency η for $\tau=0.3$ with respect to β_{r-c} for three cases of β_{c-c} and with respect to β_{c-c} for three cases of β_{r-c} are shown in Fig. 6a and b, respectively. Since τ is taken to be a constant at 0.3, both Carnot and CNCA efficiencies are constant and equal to 0.7 and 0.4523, respectively. It can be seen from Fig. 6a and b that η_{mp} is higher than η_{CNCA} for the whole range of β_{r-c} and β_{c-c} values, respectively. The difference in these efficiencies is larger for smaller β_{c-c} values in Fig. 6a and for smaller β_{r-c} values in Fig. 6b. It can be seen in Fig. 6a that as β_{r-c} is increased, η_{mp} reaches to a maximum and then decreases, however η_{mpd} always increases with increasing β_{r-c} to approach an asymptotic value for all β_{c-c} values. For high values of β_{r-c} , thermal efficiency at mp conditions, η_{mp} , approaches to the CNCA efficiency, η_{CNCA} . When $\beta_{r-c}=0$, i.e. the special case 1 is

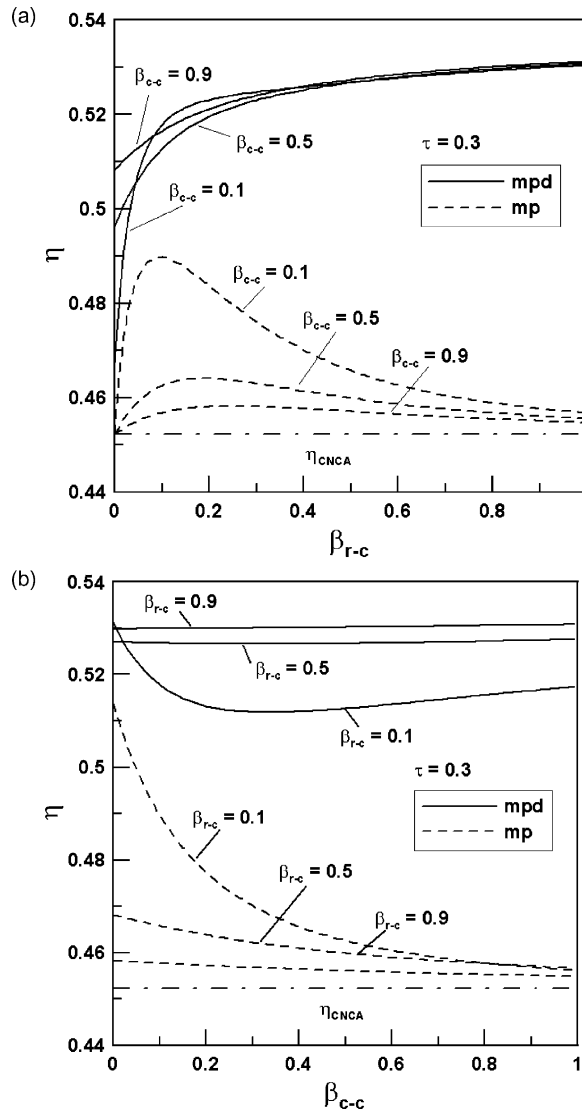


Fig. 6. Variation of efficiency η for $\tau=0.3$ (a) with respect to β_{r-c} for three cases of β_{c-c} and (b) with respect to β_{c-c} for three cases of β_{r-c} .

considered, η_{mp} is independent of β_{c-c} and has the value of η_{CNCA} being the function of the reservoir temperature ratio τ only. It can also be observed from Fig. 6b that as β_{c-c} or β_{r-c} is increased, η_{mp} always decreases and approaches to η_{CNCA} in the limit. For small values of β_{r-c} , η_{mpd} varies with β_{c-c} , however for large values of β_{r-c} , it is independent of β_{c-c} .

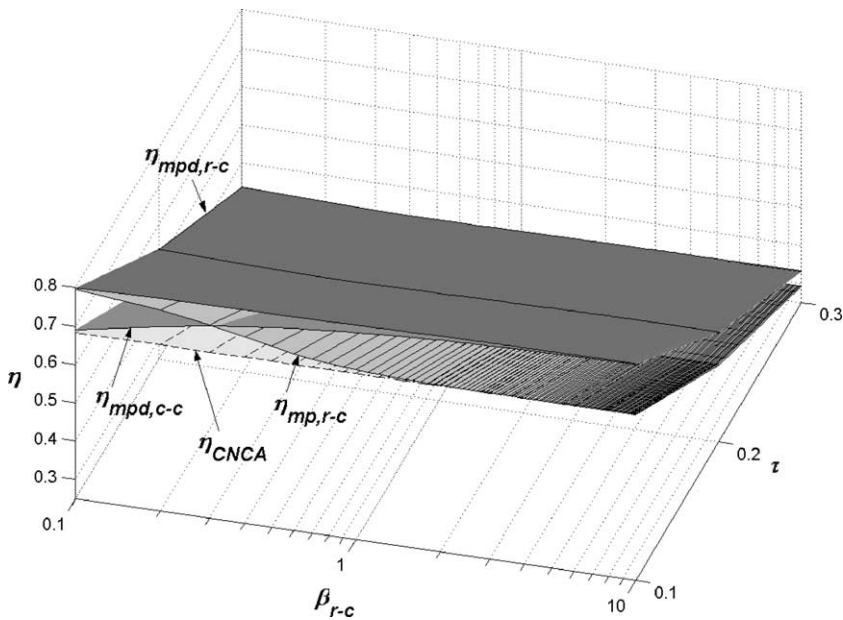


Fig. 7. Variation of efficiency η for the solar driven Carnot heat engine working at mp and mpd conditions with respect to the allocation parameter of β_{r-c} and the temperature ratio τ .

It is appropriate now to study the special case 2, i.e. when $\beta_{c-c}=0$, in order to investigate the influence of radiation only heat transfer from high temperature thermal reservoir to the heat engine. The variations of efficiencies η for the solar driven Carnot heat engine working at mp and mpd conditions with respect to the allocation parameter of β_{r-c} and τ are illustrated in Fig. 7 in order to provide a perspective for the general trends of optimal efficiencies. $\eta_{mp,r-c}$ is the efficiency at mp for the heat engine considering radiation heat transfer from the high temperature thermal reservoir and convection heat transfer to the low temperature thermal reservoir. $\eta_{mp,c-c}$ is the efficiency at maximum power for the heat engine considering heat transfers through convection at both hot and cold side thermal reservoirs, which is also called the CNCA efficiency η_{CNCA} . The efficiency for the solar driven heat engine working at mpd conditions considering heat transfer through convection at both hot and cold side thermal reservoirs $\eta_{mpd,c-c}$ is also included for comparison purposes.

To determine the effect of β_{r-c} and τ separately on the optimal efficiencies, the variations of efficiencies with respect to β_{r-c} for constant value of $\tau=0.3$ and with respect to τ for constant value of $\beta_{r-c}=0.1$ are illustrated in Figs. 8 and 9. Carnot efficiency η_C is also included for comparison. The comparisons of the efficiencies $\eta_{mp,r-c}$ and $\eta_{mp,c-c}$ in Fig. 8 may be summarized as follows: $\eta_{mp,r-c}$ is higher than $\eta_{mp,c-c}$ for the whole range of β_{r-c} values, $\eta_{mp,r-c}$ approaches to 0.538 and $\eta_{mp,c-c}$ is fixed at 0.452 therefore, the difference between these efficiencies is significant when $\beta_{r-c} \rightarrow 0$, however, this difference is very small for high values of β_{r-c} . The efficiency

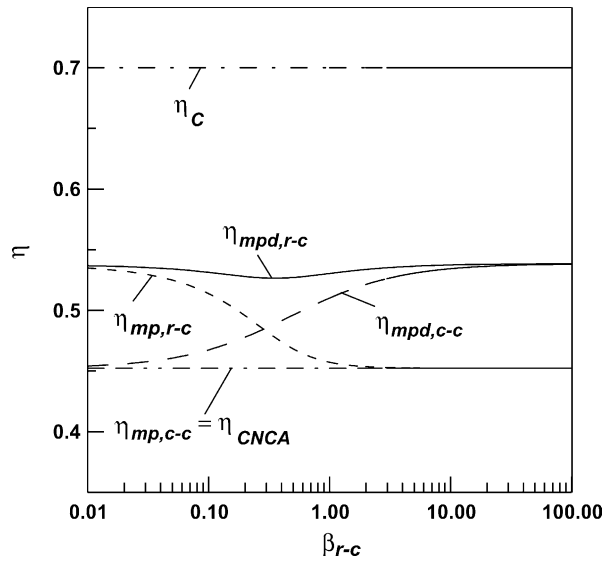


Fig. 8. Variations of efficiencies with respect to the allocation parameter β_{r-c} for constant value of $\tau=0.3$.

for the solar driven heat engine working at mpd conditions considering heat transfer through convection at both hot and cold side thermal reservoirs $\eta_{mpd,c-c}$ approaches to $\eta_{mp,c-c}$ for small values of β_{r-c} whereas it approaches to $1-2\tau/(1+\tau)$ for higher values of β_{r-c} as reported in [16]. Now, the main point of the discussion is that

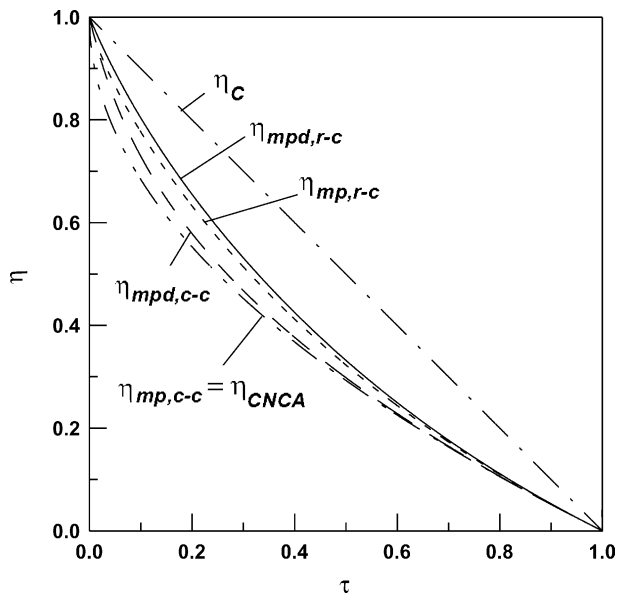


Fig. 9. Variations of efficiencies with respect to τ for constant value of $\beta_{r-c}=0.1$.

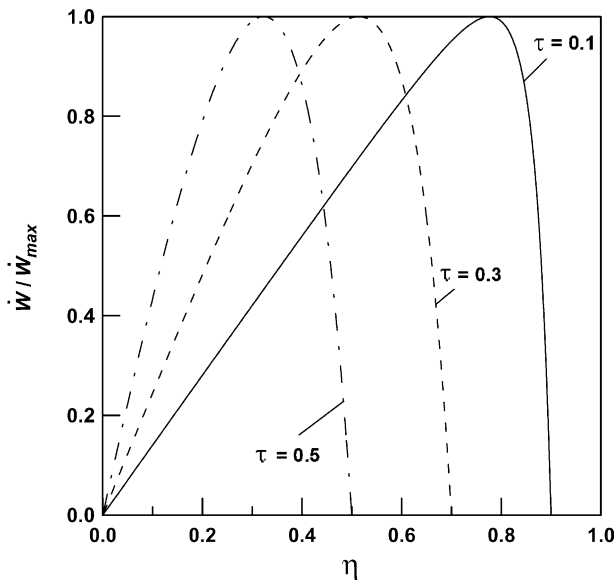


Fig. 10. Variations of normalized power $\dot{W}/\dot{W}_{\max}$ with respect to thermal efficiency η for constant value of $\beta_{r-c}=0.1$.

the efficiency at maximum power density for the heat engine considering radiation heat transfer from the high temperature thermal reservoir and convection heat transfer to the low temperature thermal reservoir $\eta_{\text{mpd},r-c}$ is always higher than $\eta_{\text{mp},r-c}$ and $\eta_{\text{mpd},c-c}$ for the whole range of β_{r-c} . It is affected by the characteristics of both heat engines mentioned above, i.e. it approaches to $\eta_{\text{mp},r-c}$ for small values of β_{r-c} , shows a small decrease at $\beta_{r-c} \cong 0.5$ and approaches to $\eta_{\text{mpd},c-c}$ for higher values of β_{r-c} . It can be seen in Fig. 9 that the efficiencies related to the solar driven Carnot heat engine having finite temperature difference $\eta_{\text{mp},r-c}$, $\eta_{\text{mpd},r-c}$ have higher values than those of Carnot heat engine having finite temperature difference $\eta_{\text{mpd},c-c}$ for the whole range of τ values. Also, the efficiencies of the heat engines working at mpd conditions are greater than those of working at mp conditions. It can be summarized that $(\eta_{\text{CNCA}} = \eta_{\text{mp},c-c}) < \eta_{\text{mpd},c-c} < \eta_{\text{mp},r-c} < \eta_{\text{mpd},r-c} < \eta_C$ for the whole range of τ values except when $\tau=0$ and $\tau=1$ where all efficiencies lead to the same value of 1 and 0, respectively.

For the special case 2, the variations of normalized power $\dot{W}/\dot{W}_{\max}$ and normalized power density $\dot{W}_d/\dot{W}_{d,\max}$ with respect to thermal efficiency η for constant value of allocation parameter $\beta_{r-c}=0.1$ are also shown in Figs. 10 and 11, respectively. It can be observed from these figures that the thermal efficiencies at mp and mpd conditions increase as τ decreases. Also, the thermal efficiency at mpd conditions yields higher values than that of at mp conditions for each τ value considered. The discussion for the special case 2 regarding the Figs. 8–11 was attempted in an earlier publication [20] however, it was lacking detail and correct interpretation of the phenomenon.

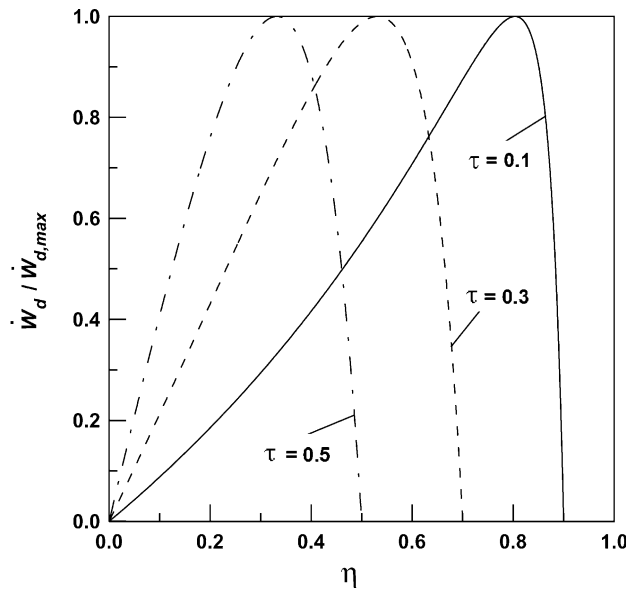


Fig. 11. Variations of normalized power density $\dot{W}_d / \dot{W}_{d,max}$ with respect to thermal efficiency η for constant value of $\beta_{r-c} = 0.1$.

4. Conclusions

A parabolic trough solar driven Rankine cycle power plant with DSG is modified to an equivalent Carnot-like cycle in order to carry out an optimal performance analysis. The performance optimization is performed for such a Carnot-like heat engine with finite-rate heat transfer working at mp and mpd conditions considering radiation only and simultaneous radiation-convection heat transfer from the high temperature thermal reservoir to the working fluid together with convection heat transfer from the working fluid to the low temperature thermal reservoir.

Two different generalized performance functions are derived to provide the optimum design parameters at mp and mpd conditions for such a solar driven heat engine having simultaneous radiation-convection heat transfer from the high temperature thermal reservoir. Two special cases are studied in detail to assess the effect of convection and radiation heat transfer mechanisms from the high temperature thermal reservoir separately.

Comparisons for the performance of such a solar driven heat engine with that of a variety of heat engines are provided. This investigation proved that the thermal efficiency of a solar driven Carnot-like heat engine at mpd conditions has the characteristics of both a solar driven Carnot heat engine at mp conditions as the allocation parameter β_{r-c} takes small values and a Carnot heat engine at mpd conditions as β_{r-c} takes large values. It is also concluded that the thermal efficiency of a solar driven Carnot-like heat engine at mpd conditions is always superior to the one at mp conditions and to the thermal efficiency of

a Carnot heat engine having finite-rate convective heat transfer with both of the thermal reservoirs.

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